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Variable Universe Fuzzy Controller for an Independent Metering System of Construction Machinery

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Abstract: As a new type of hydraulic system, the independent valve control system (IMCS) has higher flexibility compared with the traditional four-side slide valve control system. The independent metering control system (IMCS) is divided into four quadrant modes, based on the speed and load of the actuator. The traditional control methods always use a discrete control strategy to realize the switching between different modes based on mode recognition. However, the spool signals frequently switch, and the actuator operating pressure and speed are not stable when the IMCS is switched between different modes. To solve the above problems, this paper proposes a continuous control strategy for the IMCS. Based on the fuzzy control principle, the variable universe fuzzy controller (VUFC) was designed. The controller contains 49 fuzzy rules, and the output of the controller at any time is calculated by the weighted sum of several fuzzy rules. The VUFC divides the system into 49 rules, which are activated by multiple modes at any time to achieve a smooth transition between different modes. The VUFC was verified in a 37 ton excavator model and compared with the traditional independent metering meter-out (IM_MO) control strategy. The results showed that the VUFC significantly improved the frequent switching of the spool signals, and the running speed and pressure stability of the hydraulic cylinder were significantly improved. At the same time, the response speed of the hydraulic cylinder became faster at the start of the system. The VUFC can realize the continuous control of the IMCS and improve the operating performance of excavators.

Keywords: independent metering control system; variable universe fuzzy controller; smooth switching; continuous control

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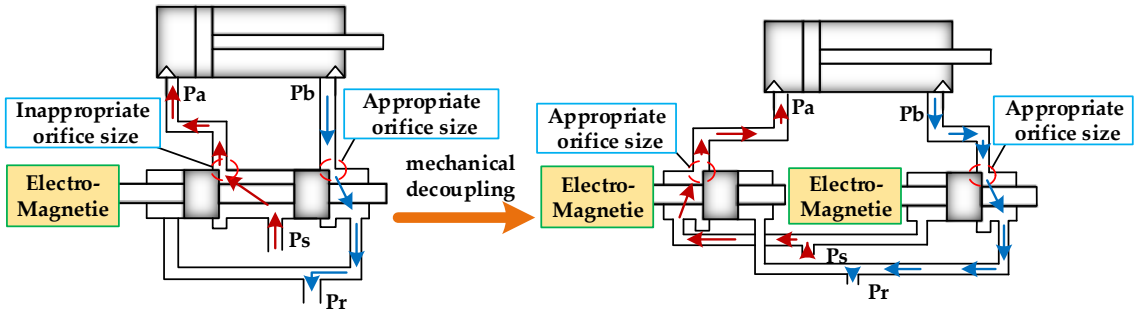
1. Introduction

As a kind of working and moving machine driven by hydraulic power, excavators have the advantages of high energy-to-weight ratio, high load capacity, high robustness and flexible operation [1]. They are widely used in harsh working environments such as earth excavation, river dredging, ore mining, emergency rescue [2]. Since there is a large demand for excavators in mining, construction, forestry and other industries [3], improving the operating performance of excavators and reducing fuel consumption and exhaust emissions are of great significance to environmental protection [4], energy conservation and national economic development [5].

In the traditional excavator hydraulic system, four side slide valves are mainly used as the control element during throttling and reversing. The element only needs to input one control signal to realize the control of actuator velocity and displacement. This solution has the advantages of simple control and strong robustness [6]. However, due to the mechanical coupling of the four side slide valves, it can only accurately control the flow and pressure at the inlet or at the outlet side of the actuator. When the excavator works, it usually has a large load and frequently changes direction. The use of traditional slide valves will lead to

excessive throttling energy loss. Therefore, the IMCS shown in Figure 1b is used instead of the traditional slide valve control system shown in Figure 1a [7]. More valves are used to control the inlet and outlet of the actuator independently [8]. This system can achieve an accurate control of the flow and pressure of the two chambers of the hydraulic cylinder simultaneously [9,10]. Compared with the traditional slide-valve control system, it has higher flexibility and significantly reduces the throttling energy consumption.



(a) Traditional control system (b) Independent metering control system

Figure 1. Mechanical decoupling principle of the independent throttling system.

The IMCS has multiple valves controlling the actuator [11] to realize the independent control of the oil inlet and outlet [12]. It realizes the accurate control of flow and pressure in the piston side and piston rod side of the hydraulic cylinder [13]. The robust adaptive controller designed by Litong Lyu [14] was applied to the IMCS composed of four proportional valves, four on–off valves, a variable displacement pump and a hydraulic cylinder, and the test showed that the system had a higher energy efficiency than the traditional one. E. Zaev [15] designed an IMCS with three crossport valves. The third valve was used to improve the active damping performance of the system. The simulation showed that the new system and control strategy improved the power utilization by 50%. Quan [16,17] proposed the velocity and position compound control strategy, which was successfully applied to a two proportional-directional-valve IMCS and passed the tests.

The IMCS has more components, which makes the hydraulic system more complex [18]. Therefore, more complex control strategies need to be designed to meet its control requirements [19]. Especially, for the moment when the slide valve switches between different modes, the collected pressure and displacement signals fluctuate near the threshold, which will cause the frequent switching of the valve between different modes, resulting in an unstable running velocity of the hydraulic cylinder [20].

In order to solve the problems of lack of smoothness and instability, in the switching process of the IMCS between different modes, M. Linjama [21,22] controlled the ramp change of the valve signals to make the spool move gently to cope with the sudden change of the operating conditions, but this reduced the dynamic performance of the system. Kim Heybroek [23,24] solved the mode-switching problem of the IMCS with an electric proportional variable pump and four proportional poppet valves but did not consider the optimal dwell time. After an in-depth investigation into the stability of the system during operating condition switching, S.F. Graebe and A. Ahlen [25] proposed a bidirectional latent tracking control strategy, which can be widely applied for a smooth operating mode switching. P.J. Antsaklis [26] studied the "dwell-time switch" of the system and calculated the optimal "dwell time" using the Lyapunov theory, simultaneously meeting the requirements of stability and response speed during the switching process. Xu et al. [27] proposed a bumpless switching method for the IMCS based on the theory of dynamic switching dwell time and bidirectional potential tracking loop. This method is a discrete control strategy A. Shenouda [28] designed a four-valve configuration to control an asymmetric hydraulic cylinder. By strictly controlling the timing sequence of four valve switches, the four valves coordinate and cooperate to realize a smooth control of

mode switching and realize a continuous control of the system in the whole working cycle. However, this method is only applicable to four- or five-valve IMCS.

Inspired by the above discrete control strategy, this paper proposes an IMCS continuous control method based on fuzzy control to avoid the switching process between different modes and improve the problem of the actuator's unstable operation in the process of discrete switching. In the process of designing a VUFC in this paper, the actuator displacement deviation and its rate of change were divided into seven states, i.e., NB, NM, NS, ZO, PS, PM and PB. Therefore, the VUFC contained 49 fuzzy rules. Several fuzzy rules took effect at any moment of the system operation, and the output was determined by the weighted summation of each fuzzy rule. Each rule in this paper corresponds to a mode of the system in the traditional discrete control strategy. That is, the smooth transition of the mode switching process could be realized through the comprehensive action of multiple modes in the system at the same time, and the discrete system could be transformed into a continuous system. The novelty presented in this paper is a VUFC that can realize the continuous control of a two-valve IMCS. The VUFC was applied to the simulation of an excavator model. The results showed that the operating performance of the hydraulic cylinder was improved, especially at the start of the machine.

The rest of this paper has the following structure. In Section 2, the IMCS and its application in an excavator are described. In Section 3, problems and related solutions of the discrete switching controller presented to the IMCS are described. In Section 4, a VUFC based on fuzzy reasoning applied to the excavator IMCS is designed. In Section 5, the VUFC is applied to the simulation of the model of excavator to verify its control performance. In Section 6, conclusions and future work are presented.

2. Studied System and Its Application in an Excavator

2.1. Application of the IMCS in an Excavator

Taking the 37 ton hydraulic excavator shown in Figure 2 as the research object, this work mainly studied the control of boom hydraulic cylinders. As shown in Figure 3, the boom independent metering control system of the 37 ton excavator was mainly composed of an electric proportional pump, two proportional directional valves, two hydraulic cylinders, sensors and other components. In this system, when valve 1 inputs negative signals and valve 2 inputs positive signals, the variable displacement pump is connected to the piston side of the hydraulic cylinder, the oil tank is connected to the piston rod side, the hydraulic cylinder extends, and the boom of the excavator is lifted. When valve 1 inputs positive signals and valve 2 inputs negative signals, the variable displacement pump is connected to the piston rod side of the hydraulic cylinder, the oil tank is connected to the piston side, the hydraulic cylinder is retracted, and the boom of the excavator is lowered.

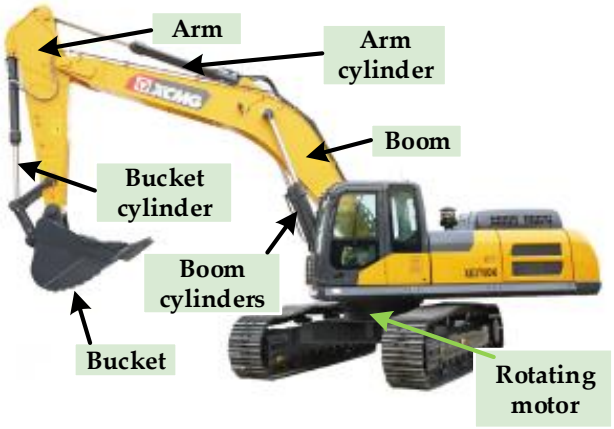


Figure 2. 37 tons hydraulic excavator.

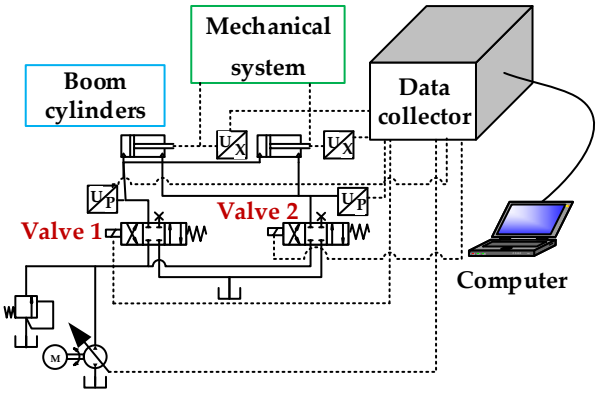


Figure 3. IMCS of the boom.

2.2. Models of Components in the IMCS

The transfer function of the spool displacement to control the current is given in Equation (1):

$$\frac{X_V(s)}{i(s)} = \frac{k_n \omega_n^2}{s^2 + 2\zeta_n \omega_n s + \omega_n^2} \tag{1}$$

where $X_V(s)$ is the spool displacement, $i(s)$ is the control current of the input valve, ω_n is the natural frequency of the valve, ζ_n is the damping coefficient of the valve, k_n is the proportional coefficient.

The transfer function of the piston displacement to the spool displacement is given in Equation (2):

$$\frac{Y(s)}{X_V(s)} = \frac{\frac{K_q}{A_h}}{s\left(\frac{s^2}{\omega_h^2} + \frac{2\zeta_h}{\omega_h}s + 1\right)} \tag{2}$$

where $Y(s)$ is the displacement of the hydraulic cylinder piston, A_h is the stress area on the piston side, ω_h is the natural frequency of the hydraulic system, ζ_h is the damping ratio of the hydraulic system, K_q is the spool valve flow gain.

The flow of oil in and out of the two chambers of the hydraulic cylinder is expressed as:

$$Q_1 = C_d w x_v \sqrt{\frac{2}{\rho}|P_s - P_1|} \tag{3}$$

$$Q_2 = C_d w x_v \sqrt{\frac{2}{\rho}|P_2 - P_t|} \tag{4}$$

where Q_1 is the piston-side inflow/outflow flow, Q_2 is the piston rod-side inflow/outflow flow, C_d is the flow coefficient, w is the area gradient, P_s is the supply pressure of the pump, P_1 is the piston-side pressure of the hydraulic cylinder, P_2 is the piston rod-side pressure, P_t is the fluid pressure in the return tank line, ρ is the hydraulic oil density.

The force balance equation of hydraulic cylinder is:

$$P_1 A_1 - P_2 A_2 = m\ddot{y} + b\dot{y} + ky + F_L \tag{5}$$

where A_1 is the force area on the piston side of the hydraulic cylinder, A_2 is the stress area on the piston rod side of the hydraulic cylinder, m is the total mass of piston and load, b is the viscous damping coefficient, k is the load spring stiffness, F_L is the external load force.

The flow continuity equation was applied to the two chambers of the hydraulic cylinder as follows:

$$\frac{V_1}{\beta_e}\dot{P}_1 = Q_1 - A_1\dot{y} \tag{6}$$

$$\frac{V_2}{\beta_e}\dot{P}_2 = A_2\dot{y} - Q_2 \tag{7}$$

where β_e is the effective bulk modulus of hydraulic oil, V_1 is the piston-side volume, V_2 is the piston rod-side volume.

3. Problem Statement

In previous studies, according to the velocity direction and force direction of the piston of the hydraulic cylinder, its working state was divided into four modes, as shown in Figure 4. From the first quadrant to the fourth quadrant, overrunning extension, resistive retraction, overrunning retraction and resistive extension are shown. Taking the work of the excavator boom as an example, the load on the boom hydraulic cylinder is mainly the total weight of the mechanical arm and the goods in the bucket. Its direction is the retraction direction of the hydraulic cylinder. Therefore, the working mode of the boom hydraulic cylinder is mainly in the third and fourth quadrants. When the boom cylinder is extended, the boom is lifted. At this time, the working mode is resistive extension. When the boom cylinder is retracted, the boom is lowered. At this time, the working mode is overrunning retraction.

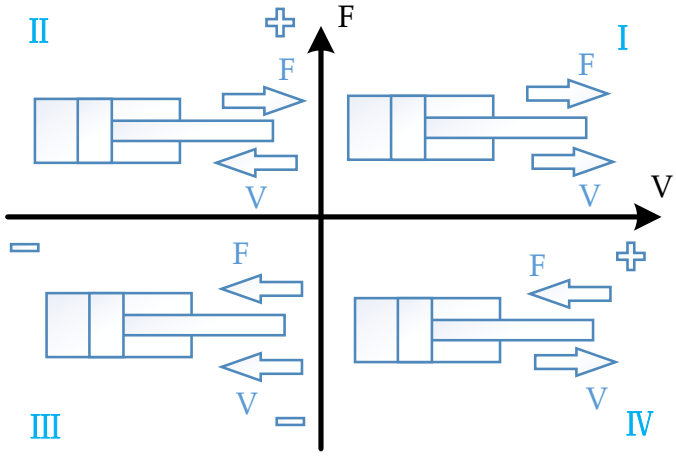


Figure 4. Working modes of the described hydraulic cylinder.

In one working cycle of the excavator, after the bucket excavates the soil, the boom is lifted and then stopped. During this process, the rotating motor makes the cab and mechanical arm rotate. After reaching the unloading position, the bucket hydraulic cylinder retracts. After unloading, the rotating motor works in reverse to make the cab and mechanical arm rotate back to the original excavation position. This process is accompanied by the boom lowering operation. During the whole excavation process, the boom hydraulic cylinder first extends, then stops for tens of seconds and then retracts to the initial position. Therefore, there are three modes in the working process of the boom cylinder: resistive extension, stopping, and overrunning retraction. The three modes are switched in in turn. A discrete control strategy needs to design corresponding controllers according to different working modes and switch to the corresponding controller according to the working mode of the actuator in the working process, as shown in Figure 5. The switching between controllers is mainly based on the pressure, displacement, velocity and other relevant information fed back by the sensor. The set threshold is used to determine the mode and then switch the control mode. When the collected feedback signal fluctuates near the threshold, it will lead to an inaccurate mode recognition and frequent switching of the valve signals between different modes. The frequent switching of the solenoid valve will cause instability in the operation velocity of actuator.

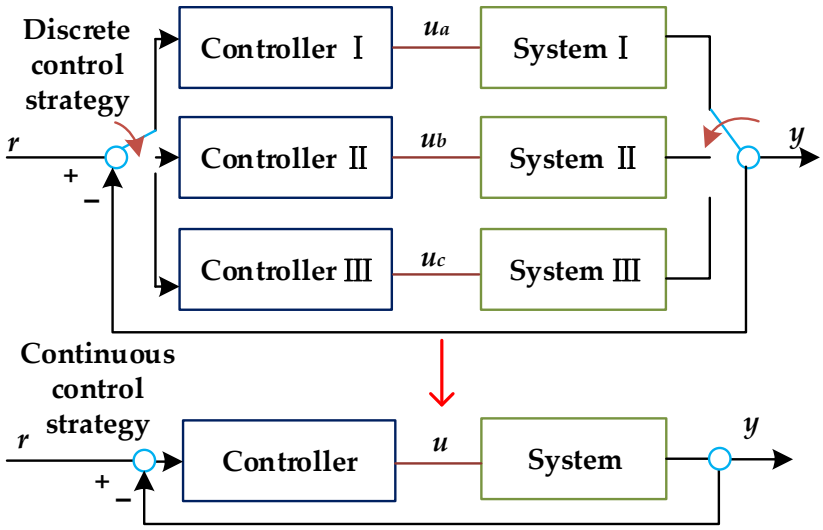


Figure 5. Motivation for a continuous control strategy.

In this paper, a variable universe fuzzy controller based on fuzzy reasoning was developed. It was applied to the continuous control of the IMCS to replace the discrete switching control strategy. It avoided the frequent switching problem of the IMCS when switching between different modes, and reduced the system pressure fluctuation.

4. Variable Universe Fuzzy Controller Design

Firstly, the fuzzy control rules were established by simulating the manual experience of skilled workers. Then, the deviation and deviation change rate were transformed into fuzzy information for fuzzy reasoning and decision making. Finally, the decision information was transformed into accurate values and input into the controlled plant.

Fuzzy control is described by an intelligent control algorithm and is a nonlinear control method. It does not depend on the accurate mathematical model of a controlled plant and has strong robustness. The hydraulic system has strong nonlinear characteristics, and it is difficult to establish an accurate model. Therefore, it is more suitable for the application of a fuzzy control algorithm. In this article, a two-input and two-output variable universe fuzzy controller was used to realize the coordinated control of two electric proportional valves in the IMCS. The hydraulic cylinder was continuously controlled throughout the movement process. The working modes caused by different loads and movement directions of the actuator were no longer distinguished. The problem of frequent switching in the switching process between the different modes of the discrete control was avoided.

4.1. Basic Principle of the Variable Universe Fuzzy Controller Based on Fuzzy Reasoning

In this paper, the VUFC only needed one displacement sensor to obtain the displacement signal. It was compared with the control target signal to obtain the displacement deviation and deviation change rate as the input of the VUFC. As shown in Figure 6, the internal structure of the VUFC is composed of two layers of ordinary fuzzy controller. The input signal of the fuzzy controller II consists of deviation and deviation change rate, and the output adjustment factor are α_1 and α_2. The displacement deviation and displacement deviation change rate were multiplied separately by α_1 and α_2. The result was used as the input signal of the fuzzy controller I. After the relevant fuzzy reasoning operation of the fuzzy controller I, the control signals of two electric proportional valves in the IMCS were obtained.

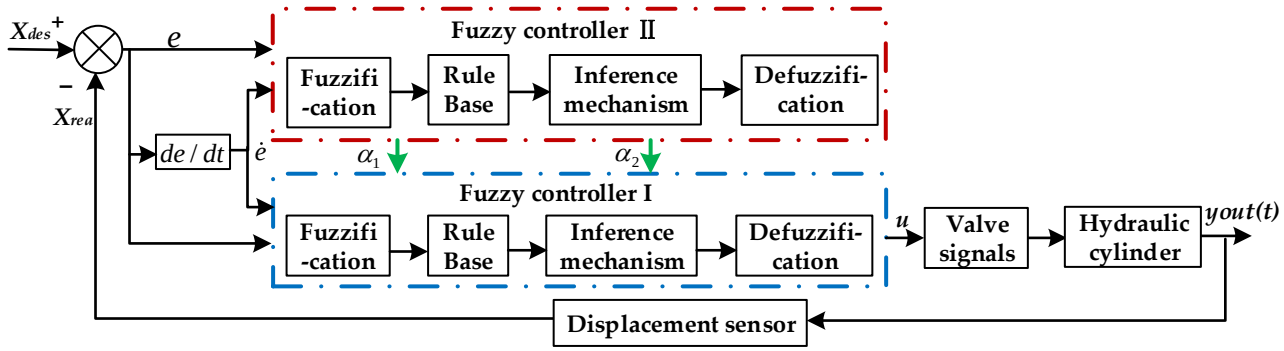


Figure 6. Block diagram of the variable universe fuzzy controller.

In this article, the mathematical expression of the variable universe fuzzy controller based on fuzzy reasoning is:

$$F(x) = \sum_{i=1}^{n} \underset{\sim}{A}(x/\alpha(x))y_i \tag{8}$$

The basic idea of the variable universe fuzzy control theory shown in Figure 7 is as follows: on the premise that the shape of rule formation remains unchanged, the universe shrinks with the decrease of deviation and extends with the increase of deviation. This process is realized by multiplying the adjustment factors separately by the deviation and its change rate. The universe shrinks in the range of small deviations, which improves the resolution and enables a finer control of the IMCS.

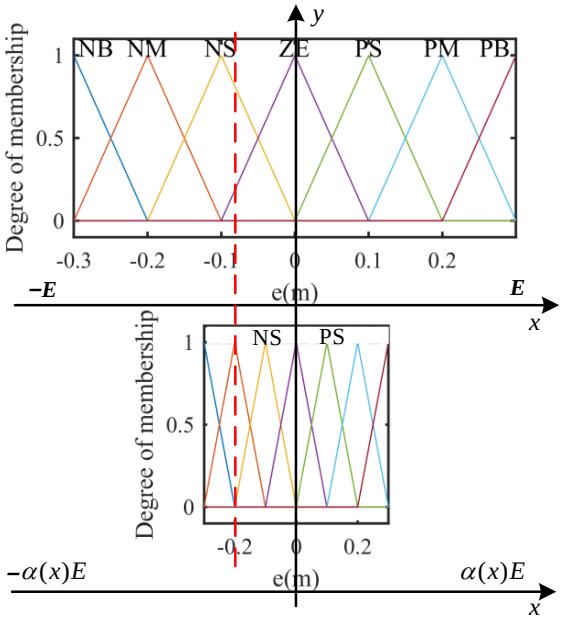


Figure 7. Basic idea of the variable universe fuzzy control.

The fuzzy subset of displacement deviation after shrinking is:

$$\mu'^{(i)}_{e} = \left(\frac{\mu_e^{(i)}(-n_e)}{-n_e\alpha_1(e)}, \frac{\mu_e^{(i)}(1-n_e)}{-(n_e-1)\alpha_1(e)}, \cdots\cdots, \frac{\mu_e^{(i)}(n_e)}{n_e\alpha_1(e)}\right) \tag{9}$$

The fuzzy subset of displacement deviation change rate after shrinking is:

$$\mu'^{(i)}_{ec} = \left(\frac{\mu_{ec}^{(i)}(-n_{ec})}{-n_{ec}\alpha_2(ec)}, \frac{\mu_{ec}^{(i)}(1-n_{ec})}{-(n_{ec}-1)\alpha_2(ec)}, \cdots\cdots, \frac{\mu_{ec}^{(i)}(n_{ec})}{n_{ec}\alpha_2(ec)}\right) \tag{10}$$

4.2. Design Process of the Variable Universe Fuzzy Controller

4.2.1. Design Process of Controller I

Firstly, the fuzzy controller I was designed, which is the base of the VUFC. Its design process drew lessons from the design method of the ordinary fuzzy controller, as shown in Figure 8.

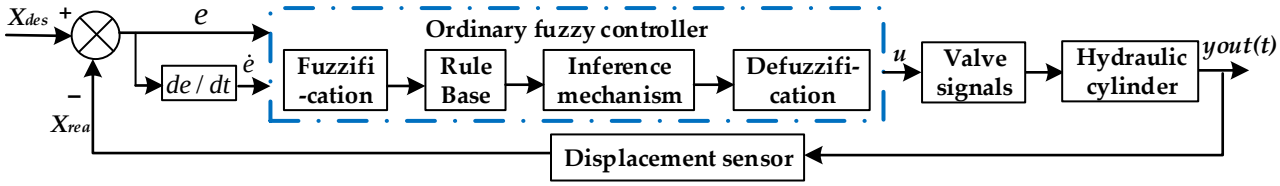


Figure 8. Block diagram of the ordinary fuzzy control system.

1. Select the word set of input and output variables

According to the experience of manual control, seven language variables are used to describe the hydraulic cylinder displacement deviation *E*, the deviation change rate *EC*, the piston-side valve control signal *U1* and the piston rod-side valve control signal *U2* separately. They are {negative large (NB), negative medium (NM), negative small (NS), zero (Z), positive small (PS), positive medium (PM), positive large (PB)}.

2. Determine the quantization factor and output scale factor

The actual range of displacement deviation and deviation change rate of the hydraulic cylinder is called the basic universe of input variables. The variables in the basic universe have exact values. According to the stroke of the boom hydraulic cylinder and the requirements of the actual work, the basic universe of displacement deviation and deviation change rate were taken as $e(t)$ $[-0.5, 0.5]$, $ec(t)$ $[-50, 50]$. The variation range of the electric proportional valve control signal was the basic universe of the output, which was $u(t)$ $[-1, 1]$. The reason for this determination was that the rated range of the control signal in the AMESim electric proportional valve model was −1~1 mA.

For the displacement deviation, the universe of fuzzy subset was E {−0.3, −0.2, −0.1, 0, 0.1, 0.2, 0.3}. For the deviation change rate, the universe of fuzzy subset was EC {−30, −20, −10, 0, 10, 20, 30}. For the control signals of the two electric proportional valves, the universe of the fuzzy subsets was U {−40, −26.66, −13.34, 0, 13.34, 26.66, 40}. According to the basic universe and the universe of fuzzy subsets, the quantization factor and scale factor could be determined by the following formulas:

$$K_e = \frac{E}{e} \tag{11}$$

$$K_{ec} = \frac{EC}{ec} \tag{12}$$

$$K_u = \frac{u}{U} \tag{13}$$

Substituting the corresponding data into the formula, we obtained: deviation quantization factor K_e = 0.6, the quantitative factor of deviation change rate K_{ec} = 0.6., the proportional factor of electric proportional valve control signal K_u = 1/40.

3. Determine the membership function of the fuzzy variables

For the convenience of calculation, the triangular membership function was mainly used in this paper. In order to improve the control accuracy, the Gaussian membership function was used at the edge of the universe, as shown in Figure 9.

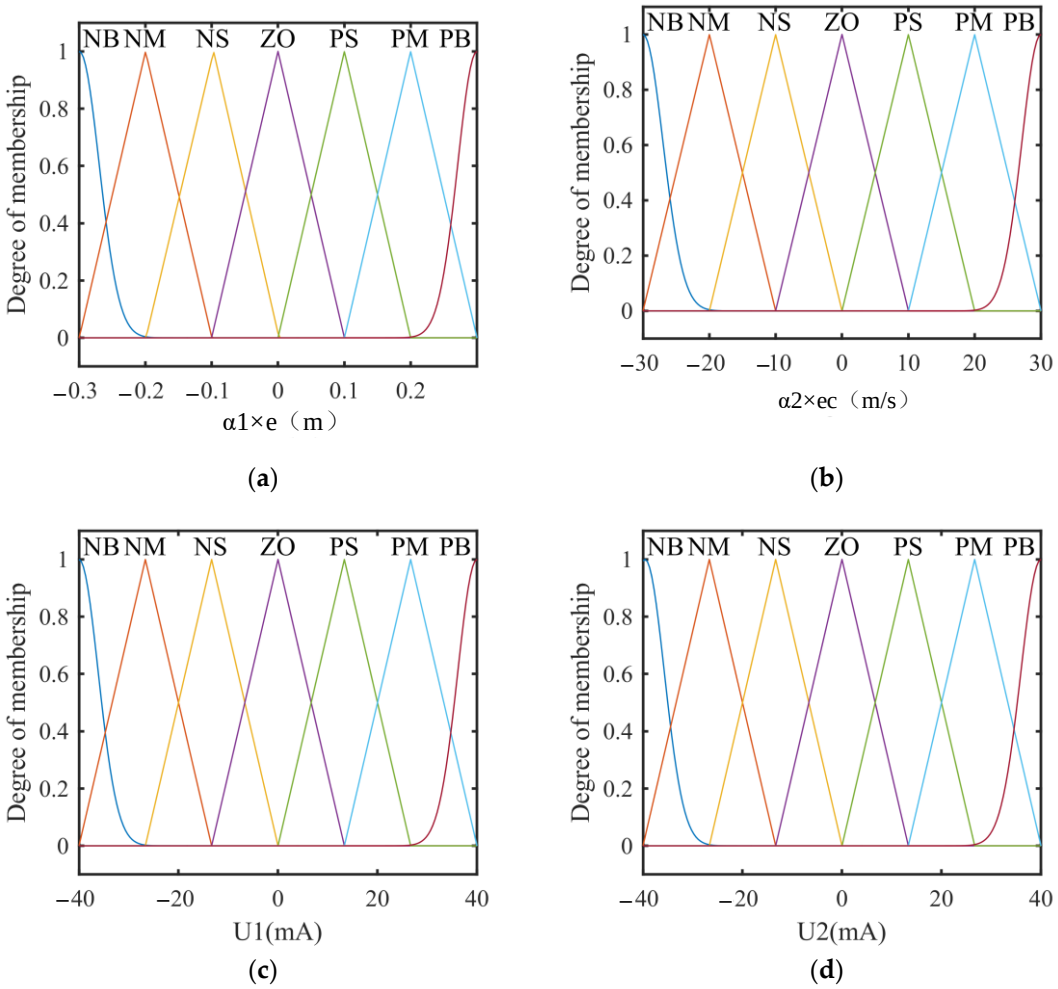


Figure 9. Membership function of the input and output variables of the fuzzy controller I. (**a**) Membership function of $\alpha 1 \times E$. (**b**) Membership function of $\alpha 2 \times EC$. (**c**) Membership function of $U1$. (**d**) Membership function of $U2$.

The analytical formula of the triangular membership function is:

$$\mu_T = \begin{cases} \frac{x-b}{a-b} & b \le x \le a \\ \frac{c-x}{c-a} & a < x \le c \\ 0 & x < b \text{ or } x > c \end{cases} \tag{14}$$

The analytic expression of the Gaussian membership function is:

$$\mu_G = \mathrm{e}^{\frac{-(x-c)^2}{2\sigma^2}} \tag{15}$$

4. Fuzzy rule design of the fuzzy controller I

The basic principles of the fuzzy control rule design are as fellows. When the values of $e(t)$ and $ec(t)$ are large, the controller needs to output a large control value $u(t)$ to reduce the deviation as soon as possible. When the values of $e(t)$ and $ec(t)$ are small, a small control value $u(t)$ is outputted to avoid overshooting. Based on the above principles and referring to previous research studies [29–31] and workers' operation experience, the fuzzy control rules were finally obtained. The fuzzy control rules of the two electric proportional valves are summarized in Tables 1 and 2, respectively.

Table 1. Fuzzy rule base of *U1*.

U1		*E*						
		NB	NM	NS	ZO	PS	PM	PB
EC	NB	PB	PB	PM	PM	PS	PS	ZO
	NM	PB	PB	PM	PS	PS	ZO	NS
	NS	PB	PM	PM	PS	ZO	NS	NM
	ZO	PM	PM	PS	ZO	NS	NM	NM
	PS	PM	PS	ZO	NS	NM	NM	NB
	PM	PS	ZO	NS	NS	NM	NB	NB
	PB	ZO	NS	NS	NM	NM	NB	NB

Table 2. Fuzzy rule base of *U2*.

U2		*E*						
		NB	NM	NS	ZO	PS	PM	PB
EC	NB	NB	NB	NB	NM	NM	NS	ZO
	NM	NB	NB	NM	NM	NS	ZO	PS
	NS	NM	NM	NM	NS	ZO	PS	PS
	ZO	NM	NS	NS	ZO	PS	PS	PM
	PS	NS	NS	ZO	PS	PM	PM	PM
	PM	NS	ZO	PS	PM	PM	PB	PB
	PB	ZO	PS	PM	PM	PB	PB	PB

5. The mapping relationship between the input and the output of the fuzzy controller I

The mapping relationship between the input and the output of the fuzzy controller I is shown in Figure 10.

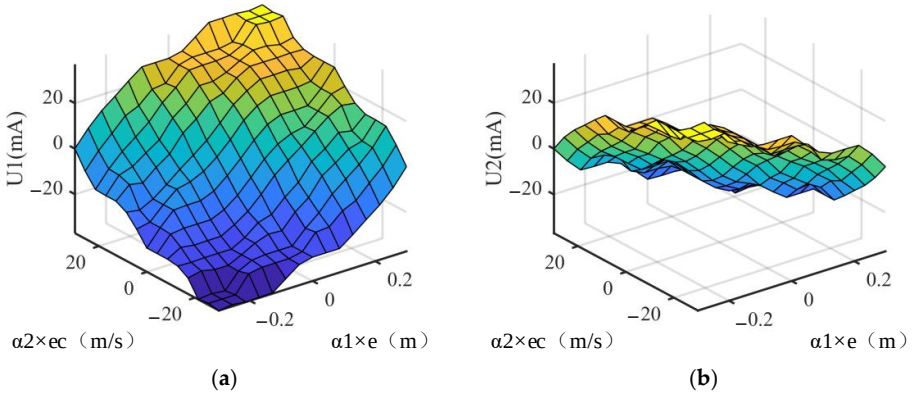


Figure 10. Mapping relationship between input variables and output variables of the fuzzy controller I. (**a**) Mapping of input variables to *U1*. (**b**) Mapping of input variables to *U2*.

4.2.2. Design Process of Controller II

In the block diagram of the variable universe fuzzy controller in Figure 7, the design process of the fuzzy controller II is the same as that of the fuzzy controller I. We referred to the design process of the fuzzy controller I to design the fuzzy controller II, as described below.

1. Select the word set of input and output variables

The language variable description of input displacement deviation E and deviation change rate EC were the same as that for controller I. They were {negative large (NB), negative medium (NM), negative small (NS), zero (Z), positive small (PS), positive medium (PM), positive large (PB)}. According to the control experience, the adjustment factor of displacement deviation factor α_1 and displacement deviation change rate α_2 were described with four language variables. They were {Z, S, M, B}.

2. Determine the quantization factor and output scale factor

Referring to controller I, the basic universe of displacement deviation and deviation change rate in the fuzzy controller II were taken as $e(t)$ $[-0.3, 0.3]$, $ec(t)$ $[-30, 30]$. The adjustment factor of displacement deviation and displacement deviation change rate were both taken as [0, 1].

The displacement deviation took the universe of fuzzy subset as E $\{-0.3, -0.2, -0.1, 0, 0.1, 0.2, 0.3\}$. The deviation change rate took the universe of fuzzy subset as EC $\{-30, -20, -10, 0, 10, 20, 30\}$. The peak value of the two adjustment factors was α $\{0.25, 0.5, 0.75, 1\}$. The quantization factor and scale factor could be determined by substituting the corresponding data into Formulas (11)–(13), obtaining: deviation quantization factor $K_e = 0.6$, deviation change rate quantization factor $K_{ec} = 0.6$, scale factor of adjustment factor $K_\alpha = 1$.

3. Determine the membership function of the fuzzy variables

For the convenience of calculation, the fuzzy controller II adopted a triangular membership function, as shown in Figure 11. Its function is expressed according to Equation (14).

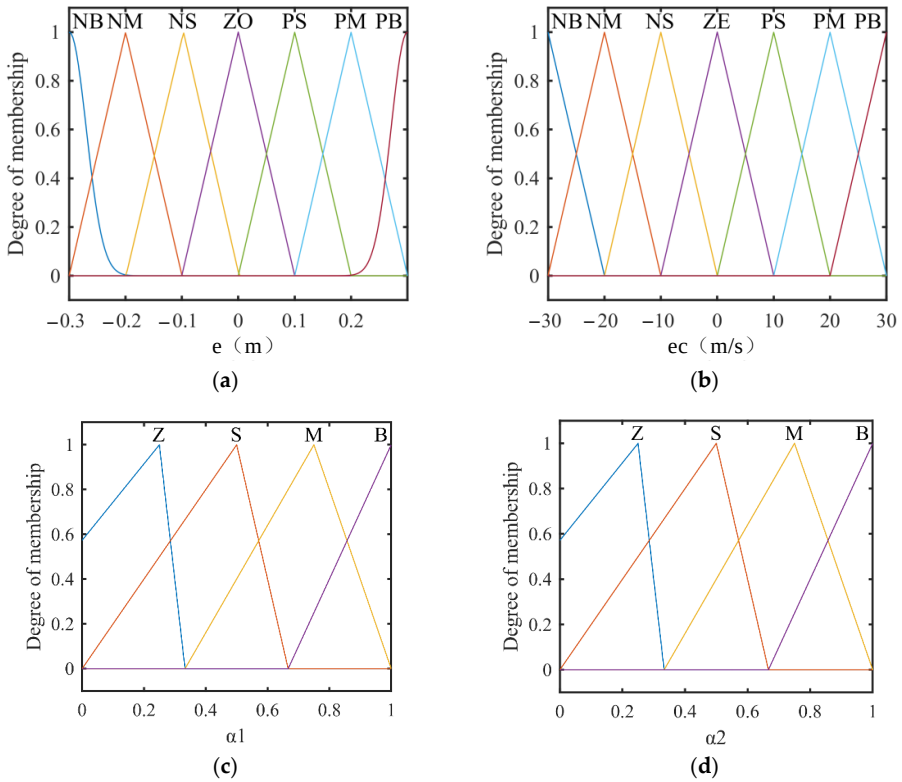


Figure 11. Membership function of input and output variables of the fuzzy controller II. (**a**) Membership function of E. (**b**) Membership function of EC. (**c**) Membership function of α_1. (**d**) Membership function of α_2.

4. Fuzzy rule design of the fuzzy controller II

The basic principles of a fuzzy control rule design is as follows. When the values of $e(t)$ and $ec(t)$ are large, the controller outputs a large adjustment factor, so that the scope

of the universe is expansion. When the values of $e(t)$ and $ec(t)$ are small, small adjustment factors are outputted to contract the universe. This improves the resolution and the control accuracy. Based on the above principles and the experience accumulated in the process of simulation and debugging, the fuzzy control rules were finally obtained. The fuzzy control rules of the two adjustment factors are summarized in Tables 3 and 4.

Table 3. Fuzzy rule base for α_1.

α_1		*E*						
		NB	NM	NS	ZO	PS	PM	PB
EC	NB	B	B	M	S	M	B	B
	NM	B	M	S	S	S	M	B
	NS	M	M	S	Z	Z	M	M
	ZO	M	S	Z	Z	Z	S	M
	PS	M	M	S	Z	S	M	M
	PM	B	M	S	S	S	M	B
	PB	B	B	M	S	M	B	B

Table 4. Fuzzy rule base for α_2.

α_2		*E*						
		NB	NM	NS	ZO	PS	PM	PB
EC	NB	B	B	M	S	M	B	B
	NM	B	M	S	S	S	M	B
	NS	M	M	S	Z	Z	M	M
	ZO	M	S	Z	Z	Z	S	M
	PS	M	M	S	Z	S	M	M
	PM	B	M	S	S	S	M	B
	PB	B	B	M	S	M	B	B

5. Mapping relationship between the input and the output of the fuzzy controller II.

The mapping relationship between the input and the output of the fuzzy controller II is shown in Figure 12.

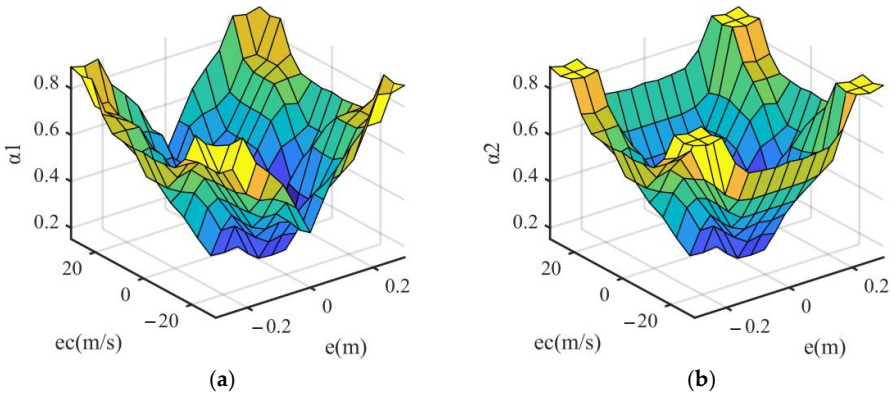


Figure 12. Mapping relationship between input variables and output variables of the fuzzy controller II. (**a**) Mapping of input variables to α_1. (**b**) Mapping of input variables to α_2.

Finally, the Simulink software was used to build the controller required for the joint simulation. The control system model is shown in Figure 13.

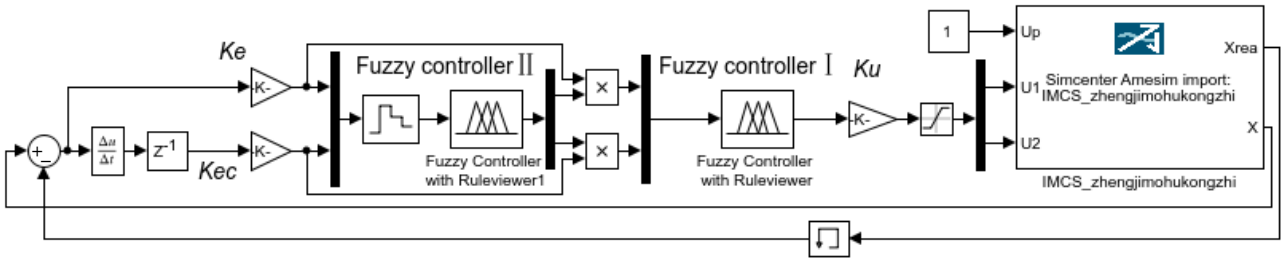


Figure 13. Simulation model of the variable universe fuzzy controller.

5. Test Research

In order to verify the feasibility of the variable universe fuzzy controller, a 37-ton excavator model was built, as shown in Figure 14. In the simulation process, the PID control strategy and variable universe fuzzy control strategy were, respectively, applied to control boom lifting, lowering and stopping. The tracking accuracy of the boom cylinder displacement, the stability of the operation velocity and the pressure fluctuation in the cylinder were compared. The superiority of the VUFC was verified.

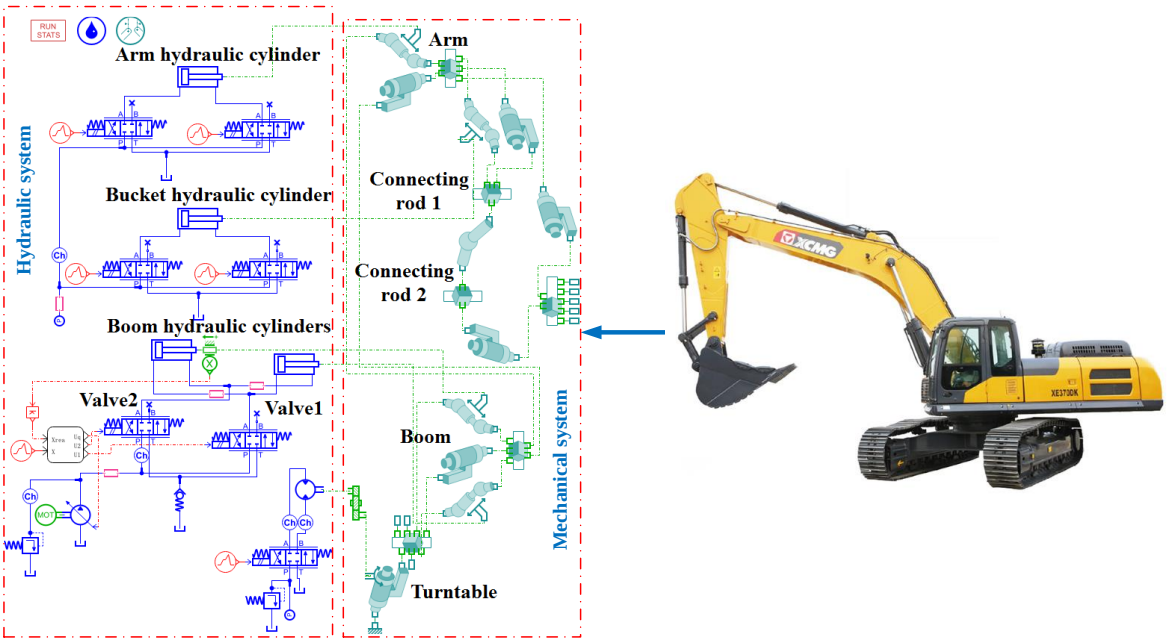


Figure 14. Image of the 37 ton IMC excavator model.

The PID control strategy is based on the independent-metering metering outlet (IMMO) principle. The corresponding control strategies based on the modes were adopted in different stages of the boom cylinder motion. In the boom lifting process, the opening of valve 1 was the largest, and valve 2 was metering. In the boom lowering process, the opening of valve 2 was the largest, and valve 1 was metering, so as to realize the control requirements of the movement process of the boom. In the positioning mode, the valve 1 signal was obtained through displacement feedback PID control. The valve 2 signal was obtained by multiplying the valve 1 signal by the corresponding area ratio. The two valves were controlled simultaneously to achieve an accurate positioning of the hydraulic cylinder.

The VUFC based on fuzzy reasoning studied in this article has two inputs and two outputs. It only controls two electric proportional valves. The input signal of the electric proportional pump was 1, which indicated the state of maximum displacement.

As shown in Figure 15, it was found that in the process of the boom start, the boom dropped to varying degrees by using the VUFC strategy and the IM-MO strategy. Of them, when the IM-MO control method was used, the boom dropped more, i.e., by 0.246 m. The descent when using the VUFC in the starting process was 0.035 m. The VUFC was more efficient than the IM-MO. When the boom cylinder reached the target position, it was obvious that the trajectory of the boom controlled by the VUFC was smoother and had higher stability.

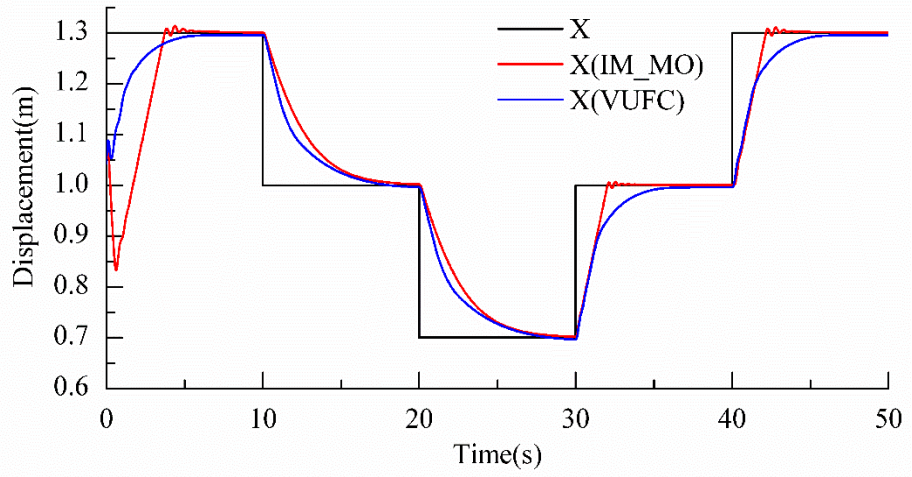


Figure 15. Boom cylinder displacement.

As shown in Figure 16, frequent switching of the valve control state occurred when using the IM-MO control strategy. In practical application, the valve cannot track such high-frequency change signals. The VUFC control strategy greatly reduced the problem of frequent spool switching during mode switching.

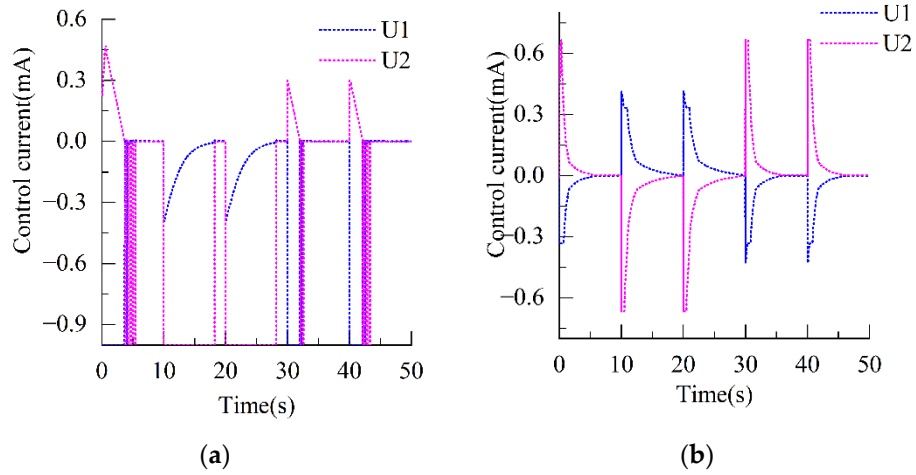


Figure 16. Signals of the electric proportional valve. (**a**) IM-MO valve signals. (**b**) VUFC valve signals.

As shown in Figure 17, the pressure when using the VUFC became basically stable and was reached faster, in 2.84 s, than when using IM-MO, which required 6.94 s, at the start of hydraulic cylinder. In addition, the pressure fluctuation amplitude of the boom lifting process was smaller. As shown in Figure 18, in the process of starting and at the first rise (0~7 s), the fluctuation velocity of the VUFC was small. It reached a stable state faster. In each boom lifting process, the VUFC could always reach a stable state first.

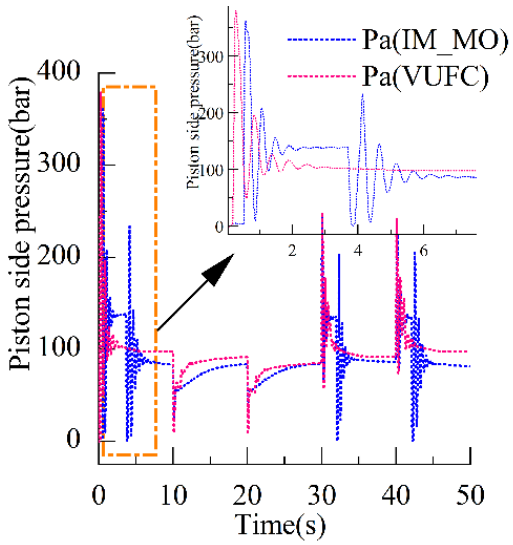


Figure 17. Piston-side pressure.

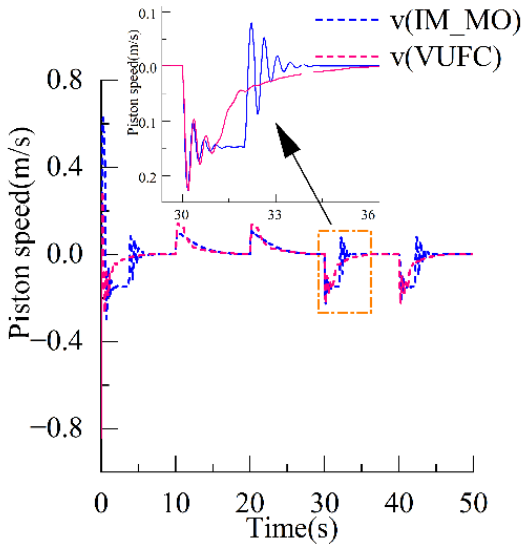


Figure 18. Velocity of the hydraulic cylinder.

As shown in Figure 19, the amplitude of the VUFC flow change was smaller than that of the IM-MO control strategy. The flow was smoother and reached a steady state faster.

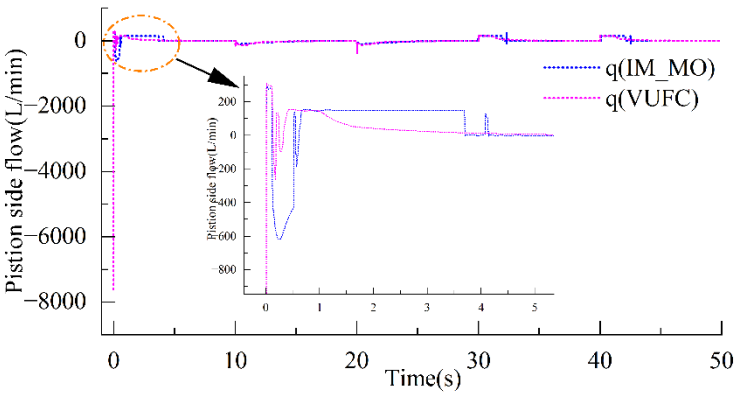


Figure 19. Piston-side flow.

6. Discussion and Conclusions

This paper developed an innovative continuous control strategy for a two-valve IMCS. It is a variable universe fuzzy controller based on fuzzy reasoning, which was applied to the IMCS of an excavator boom. The main difference between this controller and the traditional discrete controller is that the VUFC does not distinguish the working mode of the actuator. In the fuzzy controller, it refines the working mode. The weight of each fuzzy rule was calculated and summed. The control was generated by the combined action of multiple modes at any time when the actuator was running. In this way, the continuous control of the whole working cycle was realized, and the switching between different modes was avoided during the working process. By applying the IM-MO and VUFC to the model of a 37 ton IMC excavator, the boom was controlled during lifting, lowering and positioning. By comparing the hydraulic cylinder displacement tracking accuracy, pressure, speed fluctuation and other related indicators in the process of motion, the simulation results verified that the spool control signal fluctuation phenomenon was obviously improved in the process of VUFC boom starting, boom rising and boom falling. The hydraulic cylinder response speed was faster. The running speed and the system pressure were much more stable. The results showed that the VUFC control performance was more reliable.

In this paper, the VUFC did not control the proportional pump. In the next research work, the application of pump and valve collaborative control in the IMCS continuous control strategy should be considered to further reduce the system throttling energy loss. Since the fuzzy rules for the VUFC in this paper were designed according to previous experience and debugging, the best optimization cannot be guaranteed, and the intelligent algorithm will be further used to optimize the fuzzy rules, so that the controller can achieve higher performance.

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